

## COORDINATED BRAKE CONTROL SYSTEM

BACKGROUND OF THE INVENTION

[0001] The present invention relates to a coordinated brake control system of a hybrid brake system having two kinds of brake apparatuses.

[0002] U.S. Patent No. 6244674 (≈ Japanese Patent Provisional Publication No. 11-98609) discloses a coordinated brake control system for controlling a hybrid brake system constructed by a regenerative brake apparatus and a hydraulic brake apparatus. This coordinated brake control system has been arranged to improve a fuel consumption by mainly operating the regenerative brake apparatus when a vehicle speed is higher than a predetermined vehicle speed and to decrease the regenerative braking torque while increasing the hydraulic braking torque as the vehicle speed is decreased. Further, a characteristic of the regenerative braking torque to be reduced is modified in response to a pressure difference between a master-cylinder hydraulic pressure and a wheel-cylinder hydraulic pressure so as to smoothly compensate for lack of the regenerative braking torque.

SUMMARY OF THE INVENTION

[0003] However, since this coordinated brake control system has been designed without taking account of a response delay of the hydraulic brake apparatus and a response delay of the regenerative brake apparatus, it is difficult to accurately control the hydraulic braking torque and the regenerative braking torque so as to accurately correspond the sum (total braking torque) of

the regenerative braking torque and the hydraulic braking torque to a target total torque.

[0004] The applicant of the present invention has researched various methods for further smoothly varying a ratio between the regenerative braking torque and the hydraulic braking torque during a coordinated brake control while suppressing a fluctuation of the total braking torque actual value. Through the various researches, the inventors of the present invention have obtained a result that when a phase compensation is executed as to each of a hydraulic braking-torque command value and a regenerative braking-torque command value in addition to the control of U.S. Patent No. 6244674, a fluctuation of the total braking torque actual value was not improved. The inventors have found that the pressure difference between the command value and the actual value of the wheel-cylinder hydraulic pressure includes an anticipated response delay, and this anticipated response delay has been already included in the phase compensation and is different from the control error and that even if the regenerative braking torque is directly corrected, a predetermined response delay will be generated until this correction is reflected in the actual regenerative braking torque.

[0005] Further, the inventors have reached a conclusion that the generation of the response delay is caused by correcting the regenerative braking-torque command value of a high-responsibility brake such as a regenerative brake apparatus on the basis of a difference between the command value and the actual value of the hydraulic braking torque, and that this problem is solved by correcting a regenerative braking-torque command value

of a high-responsibility brake according to a difference between a reference model response value relative to the hydraulic braking-torque and an actual braking torque, where the reference model response value is obtained on  
5 the basis of a braking torque reference model which is determined upon taking account of a delay of a low-responsibility brake braking-torque relative to a torque command value of a low-responsibility brake such as a hydraulic brake apparatus.

10 [0006] It is therefore an object of the present invention to provide an improved coordinated brake control system which realizes the inventors' invention discussed above.

[0007] An aspect of the present invention resides in a  
15 coordinated brake control system for a hybrid brake system of a vehicle. The coordinated brake control system comprising: a vehicle operating condition detector detecting a vehicle operating condition of the vehicle; a first brake generating a first braking torque according  
20 to a first braking torque command value; a second brake generating a second braking torque according to a second braking torque command value, a control responsibility of the first brake being higher than a control responsibility of the second braking torque; and a  
25 controller connected to the vehicle operating condition detector, the first brake and the second brake, the controller being arranged, to determine a total braking torque command value according to the vehicle operating condition, to distribute the total braking torque command  
30 value into the first braking torque command value and the second braking torque command value, to estimate the second braking torque, to calculate a reference model

response value relative to the second braking torque command value on the basis of a braking torque reference model which is determined upon taking account of a delay of the second braking torque relative to the second

5 braking torque command value, and to correct the first braking torque command value according to a braking torque difference between the estimated second braking torque and the reference model response value.

[0008] Another aspect of the present invention resides

10 in a method of controlling a hybrid brake system of a vehicle, the hybrid brake comprising a first brake and a second brake whose control responsibility is not higher than a control responsibility of the first brake, the method comprises an operation of detecting a vehicle

15 operating condition of the vehicle; an operation of determining a total braking torque command value according to the vehicle operating condition; an operation of distributing the total braking torque command value into a first braking torque command value

20 according which the first brake generates a first braking torque and a second braking torque command value according which the second brake generates a second braking torque; an operation of estimating the second braking torque; an operation of calculating a reference

25 model response value relative to the second braking torque command value on the basis of a braking torque reference model which is determined upon taking account of a delay of the second braking torque relative to the second braking torque command value; and an operation of

30 correcting the first braking torque command value according to a braking torque difference between the

estimated second braking torque and the reference model response value.

[0009] The other objects and features of this invention will become understood from the following 5 description with reference to the accompanying drawings.

**BRIEF DESCRIPTION OF THE DRAWINGS**

[0010] Fig. 1 is a schematic view showing a complex brake provided with a coordinated brake control system according to a first embodiment of the present invention.

10 [0011] Fig. 2 is a block diagram showing controls executed by a coordinated brake controller of the coordinated brake control system shown in Fig. 1.

[0012] Fig. 3 is a flowchart showing a control program executed by the coordinated brake controller.

15 [0013] Fig. 4 is a graph showing a characteristic of an engine braking force relative to a vehicle speed when an accelerator pedal is released.

[0014] Fig. 5 is a graph showing a flat road running resistance relative to the vehicle speed.

20 [0015] Fig. 6 is a block diagram showing a deceleration controller employed in the first embodiment of the present invention.

[0016] Fig. 7 is a graph showing an allowable maximum value and a maximum limit value of a regenerative braking 25 torque relative to the vehicle speed.

[0017] Fig. 8 is a graph showing a normal distribution characteristic of the braking torque to front and rear wheels.

30 [0018] Fig. 9 is a block diagram showing a hydraulic controller for controlling a wheel cylinder hydraulic pressure in the first embodiment of the present invention.

[0018] Figs. 10A through 10F are time charts showing a

coordinated control operation in case that wheel-cylinder hydraulic control error is around zero.

[0019] Figs. 11A through 11F are time charts showing a coordinated control operation in case that the correction 5 of the regenerative braking-torque command value and the phase advance compensation are not executed even when wheel-cylinder hydraulic control error is generated.

[0020] Figs. 12A through 12F are time charts showing a coordinated control operation in case that the correction 10 of the regenerative braking-torque command value is executed, and the phase advance compensation is not executed when wheel-cylinder hydraulic control error is generated.

[0021] Figs. 13A through 13F are time charts showing a coordinated control operation in case that both of the 15 correction of the regenerative braking-torque command value and the phase advance compensation are executed when wheel-cylinder hydraulic control error is generated.

[0022] Fig. 14A is a part of a flowchart showing a newly employed steps in the control program executed by 20 the coordinated brake controller of a second embodiment according to the present invention, and Fig. 14B is a block diagram showing a control executed by the coordinated brake controller of the coordinated brake control system according to the second embodiment of the 25 present invention.

[0023] Figs. 15A through 15F are time charts showing a coordinated control operation of the hybrid brake system shown in Fig. 14B in case that a deceleration feedback 30 control is executed when wheel-cylinder hydraulic control error is generated.

[0024] Figs. 16A through 16F are time charts showing a coordinated control operation of the hybrid brake system shown in Fig. 14B in case that the deceleration feedback control, the correction of the regenerative braking-torque command value, and the phase advance compensation are executed when wheel-cylinder hydraulic control error is generated.

[0025] Figs. 17A through 17F are time charts showing a coordinated control operation of the hybrid brake system shown in Fig. 14B in case that the deceleration feedback control, the correction of the regenerative braking-torque command using a high-frequency component of the hydraulic control error, and the phase advance compensation are executed when wheel-cylinder hydraulic control error is generated.

#### DETAILED DESCRIPTION OF THE INVENTION

[0026] Referring to Figs. 1 through 13F, there is discussed a first embodiment of a coordinated brake control system for a hybrid brake system in accordance with the present invention.

[0027] Fig. 1 shows the hybrid brake system comprising the coordinated brake control system according to the first embodiment of the present invention. The hybrid brake system comprises a hydraulic brake apparatus of generating a braking force by supplying a hydraulic pressure to a wheel cylinder 2 provided for each driving wheel 1 (in Fig. 1, only one wheel is shown), and a regenerative brake apparatus of converting a wheel rotating energy into an electric power by means of an alternating-current (AC) synchronous motor 4 which is connected to driving wheel 1 through a gear box 3.

[0028] The coordinated brake control system of the hybrid brake system is arranged to effectively recover a regenerative energy by decreasing a braking hydraulic pressure to wheel cylinder 2 during when the barking torque (force) is mainly produced (covered) by the regenerative braking torque using AC synchronous motor 4.

[0029] First, there is discussed the hydraulic brake apparatus which does not exceed in response characteristic as compared with a regenerative brake apparatus. When a brake pedal 5 is depressed according to the driver's braking intent, a depression force of brake pedal 5 is amplified by a hydraulic booster 6. The amplified depression force pushes a piston cup of a master cylinder 7, and therefore mater cylinder 7 outputs a master-cylinder hydraulic pressure Pmc corresponding to the depression force of brake pedal 5 toward a brake hydraulic conduit 8. Although Fig. 1 shows that brake hydraulic conduit 8 is connected only to wheel cylinder 2 of front drive wheel 1, it is of course that brake hydraulic conduit 8 is connected to other wheel cylinders of other three wheels.

[0030] Brake fluid in a reservoir 9 is commonly used by hydraulic booster 6 and master cylinder 7 and serves as working fluid. Hydraulic booster 6 comprises a pump 10 which sucks brake fluid from reservoir 9 and discharges the brake fluid toward an accumulator 11 to store the pressurized working fluid in accumulator 11. Further, the hydraulic pressure in accumulator 11 is controlled by a sequential control using a pressure sensor 12 provided in a conduit between pump 19 and accumulator 11.

[0031] Hydraulic booster 6 amplifies the depression force applied to brake pedal 5 using the hydraulic pressure in accumulator 11 as a pressure source, and presses the piston cup of master cylinder 7 by mean of the  
5 amplified depression force. Master cylinder 7 generates master-cylinder hydraulic pressure Pmc corresponding to the brake pedal depression force by pressingly packing (containing) the brake fluid supplied from reservoir 9 in brake conduit 8, and supplies wheel-cylinder hydraulic  
10 pressure Pwc to wheel cylinder 2 as a base pressure.

[0032] Wheel-cylinder hydraulic pressure Pwc is feedback controlled using the accumulator pressure in accumulator 11, as discussed later. In order to achieve this feedback control, an electromagnetic selector valve  
15 13 is provided in brake hydraulic conduit 8, and a pressure increasing circuit 15 and a pressure decreasing circuit 17 are connected to a brake hydraulic conduit 8 at a position nearer to wheel cylinder 2 as compared with the position of electromagnetic selector valve 13.  
20 Pressure increasing circuit 15 extends from a discharging port of pump 10 and comprises a pressure increasing valve 14. Pressure decreasing circuit 17 extends from a suction port of pump 10 and comprises a pressure decreasing valve 16.

25 [0033] Electromagnetic selector valve 13 is a normal open valve, and therefore master-cylinder hydraulic pressure Pwc is increased by the pressure of accumulator 11 which is produced by the fluid communication with pressure increasing circuit 15 when electromagnetic  
30 selector value 13 is in an off state corresponding to a normal open state. When a solenoid 13a of electromagnetic selector valve 13 is energized to close

brake hydraulic circuit 8, master cylinder 7 is simultaneously communicated with a stroke simulator 26 to apply a hydraulic load corresponding (equal) to that of wheel cylinders 2 to master cylinder 7. This  
5 communication with stroke simulator 26 applies an operation feeling as same as that at the normal state to brake pedal 5.

[0034] Pressure increasing valve 14 is also a normal open valve and increases wheel-cylinder hydraulic pressure Pwc using the pressure of accumulator 11 (by the communication with pressure increasing circuit 15) when put in the norm open state (de-energized). On the other hand, when a solenoid 14a of pressure increasing valve 14 is energized, increase of wheel-cylinder hydraulic pressure Pwc is stopped by shutting off the communication between brake hydraulic conduit 8 and pressure increasing circuit 15. Pressure decreasing valve 16 is a normal close valve which is closed when a solenoid 16a is de-energized. When solenoid 16a is energized, pressure decreasing circuit 16 is communicated with brake hydraulic conduit 8 so as to decrease wheel-cylinder hydraulic pressure Pwc.  
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[0035] Pressure increasing valve 14 and pressure increasing valve 16 are put in closed state so that pressure increasing circuit 15 and pressure decreasing circuit 17 are shut off from brake hydraulic conduit 8, during when electromagnetic selector valve 13 is put in an open state so as to open brake hydraulic conduit 8. Therefore, wheel-cylinder hydraulic pressure Pwc is  
30 determined from master-cylinder hydraulic pressure Pmc. Further, during when wheel-cylinder hydraulic pressure Pwc is increased or decreased by operating pressure

increasing valve 14 or pressure decreasing valve 16, electromagnetic selector valve 13 is being turned on to shut off brake hydraulic conduit 8 so as not to be affected by master-cylinder hydraulic pressure Pmc.

5 [0036] Hydraulic brake controller 18 controls electromagnetic selector valve 13, pressure increasing valve 14 and pressure decreasing valve 16. Hydraulic brake controller 18 receives a signal indicative of master-cylinder hydraulic pressure Pmc from a pressure  
10 sensor 19 and a signal indicative of wheel-cylinder hydraulic pressure Pwc from a pressure sensor 20. Master-cylinder hydraulic pressure Pmc represents a braking torque (force) demanded by a driver, and wheel-cylinder hydraulic pressure Pwc represents an  
15 actual value of a hydraulic braking torque.

[0037] AC synchronous motor 4 is drivingly connected to each driving wheel 1 through gear box 3, and is controlled by a motor torque controller 21. Motor torque controller 21 outputs three-phase PWM signal, and  
20 inverter (DC-AC inverting current control circuit) 22 inverts DC into AC and supplies the inverted AC to AC synchronous motor 4. When a driving torque (driving force) by motor 4 is required, driving wheels 1 are driven by an electric power from a DC battery 23. When a  
25 braking torque (braking force) by motor 4 is required, a vehicle motion energy is recovered in battery 23 by executing a regenerative braking torque control.

[0038] As shown in Fig. 1, a hydraulic brake controller 18 and a motor torque controller 21 are  
30 networked with a coordinated brake controller 24. Hydraulic brake controller 18 controls the hydraulic brake apparatus according to the command signal from

coordinated brake controller 24, and motor torque controller 21 controls the regenerative braking control apparatus according to the command signal from coordinated brake controller 24. More specifically,  
5 motor torque controller 21 controls a regenerative brake torque generated by motor 4 on the basis of a regenerative braking-torque command value outputted from coordination controller 24. When driving of driving wheels 1 is required, motor torque controller 21 executes  
10 the driving torque control using motor 4. Further, motor torque controller 21 calculates an allowable maximum regenerative braking torque of motor 4, which is determined based on a charged state and a temperature of battery 23, and outputs a signal corresponding to the  
15 allowable maximum regenerative braking torque to coordinated brake controller 24.

[0039] Coordinated brake controller 24 receives the signal indicative of master-cylinder hydraulic pressure Pmc from pressure sensor 19 through hydraulic brake controller 18, the signal indicative of wheel-cylinder hydraulic pressure Pwc from pressure sensor 20 through hydraulic brake controller 18, and a signal indicative of a wheel speed Vw of driving wheels 1 from wheel speed sensors 25.

25 [0040] Coordinated brake controller 24 executes a coordination control of the hybrid brake system by executing a processing based on the above input information, as shown by a block diagram in Fig. 2 and a flowchart in Fig. 3. Fig. 3 is a timer interruption processing executed at 10msec intervals.

[0041] At step S1 controller 24 calculates (measures) master-cylinder hydraulic pressure Pmc and wheel-cylinder hydraulic pressure Pwc from the received information.

[0042] At step S2 controller 24 calculates a driving 5 wheel speed Vw based on the received signal indicative of driving wheel speed Vw. Further, controller 24 calculates a driving-wheel deceleration  $\alpha_v$  by executing a filtering processing of driving wheel speed Vw using a band-pass filter represented by the following transfer 10 function Fbpf(s).

$$F_{bpf}(s) = 1/\{(1/\omega^2) s^2 + (2\xi/\omega) s + 1\} \quad \dots \quad (1)$$

where s is Laplace operator. Actually, driving-wheel deceleration  $\alpha_v$  is calculated using a recurrence formula obtained by discretizing the transfer function expressed 15 by the expression (1) with Tustin (Biliner) approximation.

[0043] At step S3 controller 24 reads a maximum regenerative braking torque Tmmax, which motor 4 is capable of generating, from a high-speed communication buffer between controller 24 and motor torque controller 20 21. As discussed above, motor torque controller 21 calculates maximum regenerative braking torque Tmmax according to a charged state of battery 23 and the like. For example, maximum regenerative braking torque Tmmax 25 (driving wheel speed Vw) is varied according to vehicle speed VSP as shown in Fig. 7.

[0044] At step S4 controller 24 calculates a target deceleration  $\alpha_{dem}$  of the vehicle using following expression (2), master-cylinder hydraulic pressure Pmc and a constant K1 which is determined according to 30 vehicle specifications previously stored in ROM of controller 24.

$$\alpha_{dem} = -(Pmc \times K1) \quad \text{---(2)}$$

where a negative value of acceleration  $\alpha$  is a deceleration, and a negative value of torque  $T$  is a braking torque.

5 [0045] Target deceleration  $\alpha_{dem}$  is not determined only by master-cylinder hydraulic pressure  $Pmc$ , which is a physical quantity demanded by a driver. For example, if the vehicle is equipped with an inter-vehicle distance control system and/or a cruise control system, target  
10 deceleration  $\alpha_{dem}$  is determined upon taking account of a physical quantity of an automatic braking executed by the inter-vehicle distance control system and/or the cruise control system.

[0046] At step S5 controller 24 estimates a vehicle  
15 deceleration (engine brake deceleration)  $\alpha_{eng}$  which is produced only by the negative driving force generated when the accelerator pedal is released, that is, by an engine brake force.

[0047] More specifically, controller 24 retrieves an  
20 engine brake force estimated value (target engine brake force)  $Teng$  from a map shown in Fig. 4, which has been previously stored in ROM of controller 24, according to vehicle speed VSP (driving wheel speed  $Vw$ ) and a selected range (D-range or L-range) of an automatic transmission.  
25 Further, controller 24 retrieves a flat-road running resistance  $Treg$  from a map shown in Fig. 5, which has been previously stored in ROM of controller 24, according to vehicle speed VSP (driving wheel speed  $Vw$ ).  
Furthermore, controller 24 calculates engine brake  
30 deceleration estimated value  $\alpha_{eng}$ , which is an average value on a flat road, by dividing the sum of engine brake

force estimated value Teng and flat-road running resistance Treg by a vehicle weight Mv as shown by the following expression (3).

$$\alpha_{eng} = (Teng + Treg)/Mv \quad \text{---(3)}$$

5 [0048] At step S6 controller 24 calculates a braking-torque command value Tdff (a feedforward compensation quantity), which is necessary to realize target deceleration  $\alpha_{dem}$  by the following manner. That is, controller 24 converts target deceleration  $\alpha_{dem}$  to the  
 10 braking torque using constant K2 determined from the vehicle specifications. Subsequently, braking-torque command value Tdff (feedforward compensation quantity) for target deceleration  $\alpha_{dem}$  is obtained by filtering the braking torque corresponding to target deceleration  $\alpha_{dem}$   
 15 through a characteristic  $C_{FF}(s)$  of feedforward compensator (phase compensator) 51, which is represented by the following in expression (4) and functions to correspond a response characteristic  $Pm(s)$  of a controlled object (vehicle) 54 to a characteristic  $Fref(s)$  of a reference model 52 in Fig. 6. Actually, braking-torque command  
 20 value Tdff (feedforward compensation quantity) for target deceleration  $\alpha_{dem}$  is calculated by discretizing the expression (4) as is the same manner discussed above.

$$G_{FF}(s) = Fref(s)/Pm(s) \quad \text{---(4)}$$

$$25 = (Tp \cdot s + 1) / (Tr \cdot s + 1) \quad \text{---(5)}$$

where  $Tp$  is a time constant, and  $Tr$  is also a time constant.

[0049] At step S7 controller 24 determines whether or not a brake pedal operation is executed, by determining  
 30 whether or not master-cylinder hydraulic pressure  $Pmc$  is greater than or equal to a small set value. When the

determination at step S7 is affirmative, that is, when the brake pedal operation is executed, the program proceeds to step S10 wherein controller 24 determines whether or not a condition of no brake-pedal operation is continued for more than preset time tSET by determining whether or not timer tBOFF is greater than or equal to set time tSET.

[0050] When the determination at step S10 is affirmative ( $tBOFF \geq tSET$ ), that is, when the condition of no brake pedal operation continues for more than preset time tSET, the program proceeds to step S11 wherein controller 24 updates deceleration reference value  $\alpha_B$  to deceleration  $\alpha_0$ . When the determination at step S10 is negative, that is, when the condition of no brake pedal operation is not continued for more than preset time tSET, the program jumps to step S12 wherein controller 24 resets timer tBOFF at zero ( $tBOFF=0$ ).

[0051] At step S13 controller 24 calculates a total braking-torque command value necessary to achieving target deceleration  $\alpha_{dem}$  by the following manner.

[0052] A deceleration controller employed in the first embodiment according to the present invention is constructed by a two-degree-of-freedom control system, and comprises feedforward compensator 51, reference model 52 and feedback compensator 53 as shown in Fig. 6. Feedback compensator 53 achieves a close-loop performance such as the stability and the robustness of the control system, and feedforward compensator 51 achieves a responsibility to target deceleration  $\alpha_{dem}$  as far as there is no model error.

[0053] In calculation of a braking-torque feedback compensation quantity  $T_{dfb}$ , reference model response deceleration  $\alpha_{dem}$  is first obtained by filtering target deceleration  $\alpha_{dem}$  through a reference model having a characteristic  $F_{ref}(s)$  represented by the following expression (6).

$$F_{ref}(s) = 1 / (T_r \cdot s + 1) \quad \text{---(6)}$$

[0054] Further, as shown in Fig. 6, a deceleration feedback difference  $\Delta\alpha$  is obtained by subtracting a difference  $(\alpha_v - \alpha_B)$  of an actual deceleration  $\alpha_v$  and an offset quantity  $\alpha_B$  from reference model response deceleration  $\alpha_{ref}$ , as follows.

$$\Delta\alpha = \alpha_{ref} - (\alpha_v - \alpha_B) \quad \text{---(7)}$$

Furthermore, braking torque compensation quantity  $T_{dfb}$  is obtained by filtering deceleration feedback difference  $\Delta\alpha$  through feedback compensator 53 having a characteristic  $C_{FB}(s)$  represented by the following expression (8).

$$C_{FB}(s) = (K_p \cdot s + K_i) / s \quad \text{---(8)}$$

In this first embodiment, this characteristic is achieved by a basic PI controller, and therefore control constants  $K_p$  and  $K_i$  thereof are determined upon taking account of a gain margin and a phase margin. Further, characteristics  $F_{ref}(s)$  and  $C_{FB}(s)$  are obtained by discretizing the expressions (6) and (8) as is the same manner discussed above.

[0055] Subsequently, total braking-torque command value  $T_{dcom}$  is obtained by summing braking-torque command value  $T_{dff}$  (feedforward compensation quantity) for target deceleration  $\alpha_{dem}$  and braking torque feedback compensation quantity  $T_{dfb}$  ( $T_{dcom} = T_{dff} + T_{dfb}$ ), as shown in Fig. 6.

With this series of executions at step S13, total braking-torque command value  $T_{dcom}$  is obtained. Therefore, step S13 in Fig. 3 corresponds to total braking-force command value determining means 31 in Fig.

5 2.

[0056] At step S14 subsequent to the execution of step S13 or S9, controller 24 calculates a maximum regenerative braking torque limit value  $T_{mmax(Lim)}$  by executing a limiting operation of maximum regenerative 10 braking torque  $T_{mmax}$  as follows.

[0057] First, in order to ensure a margin for enabling the correction (plus and minus correction) of regenerating braking torque based on the wheel-cylinder hydraulic pressure control error, limit value  $T_{mmax(Lim)}$  15 of the maximum regenerative braking torque is set at 80% of maximum regenerative braking torque  $T_{mmax}$  ( $T_{mmax(Lim)}=0.8\times T_{mmax}$ ), as shown by a broken line in Fig. 7.

[0058] Further, in order to accomplish the 20 regenerative coordinated brake control by smoothly varying a weighted ratio state of the regenerative braking and the hydraulic braking from a regenerative braking weighted state to a hydraulic braking weighted state, limit value  $T_{mmax(Lim)}$  of the maximum regenerative 25 braking torque is set at a value represented by a bold line in Fig. 7 so as to gradually decrease a magnitude of limit value  $T_{mmax(Lim)}$  toward zero as vehicle speed VSP (driving wheel speed  $V_w$ ) decreases and to finally take zero, as shown in Fig. 7. Therefore, step S14 in Fig. 3 30 corresponds to maximum regenerative braking torque limiting means 32 in Fig. 2.

[0059] At step S15 controller 24 distributes total braking-torque command value Tdcom for regenerative coordination brake control into regenerative braking torque command Tmcom and hydraulic braking-torque command 5 value Tbcom, based on maximum regenerative braking torque limit value Tmmax(Lim). Therefore, step S15 corresponds to distributing hydraulic braking torque and regenerative braking torque means 33 in Fig. 2.

[0060] In this first embodiment according to the 10 present invention, the distribution thereof is executed so that regenerative braking torque command Tmcom takes maximum regenerative braking torque limit value Tmmax(Lim) as possible, that is, so as to consume maximum regenerative braking torque 15 limit value Tmmax(Lim) as possible. Further, hydraulic braking-torque command value Tbcom is distributed to the front wheel (driving wheel) side and the rear wheel (driven wheel) side. Further, since the first embodiment has been shown and described such that motor 4 for 20 regenerative braking is provided only to front wheels 1 acting as driving wheels, there are occurred Mode 1 and Mode 2 where normal front and rear braking-torque distribution is maintained and Mode 3 and Mode 4 where normal front and rear distribution of braking torque 25 cannot be maintained.

[0061] First, normal front-wheel braking-torque command value Tdcomf and the normal rear-wheel braking-torque command value Tdcomr are obtained by normally distributing total braking-torque command value 30 Tdcom to the front wheel side and the rear wheel side on the basis of the map data shown in Fig. 8 which has been previously stored in ROM of controller 24. The normal

front and rear braking torque distribution is a front and rear braking force (torque) distribution characteristic which is a reference value when the regenerative braking is not executed, and has been determined upon taking

5 account of the rear-wheel lock avoidance, the stability of the vehicle behavior, and the shortening of the braking distance which are caused by the weight movement between front and rear wheels during the braking operation.

10 [0062] Hereinafter, the regenerative coordination braking control by each Mode as follows.

[Mode 4]

When  $T_{mmax}(Lim) \leq (Tdcomf + Tdcomr)$ , only the regenerative braking is employed as expressed by the  
15 following expressions (9A):

$$Tbcomf = 0,$$

$$Tbcomr = 0, \text{ and}$$

$$Tmcom = Tdcomf + Tdcomr. \quad \text{---(9A)}$$

[Mode 3]

20 When  $T_{mmax}(Lim) \leq Tdcomf$ , the regenerative braking and the rear wheel hydraulic braking are employed as expressed by the following expressions (9B):

$$Tbcomf = 0$$

$$Tbcomr = Tdcomf + Tdcomr - T_{mmax}(Lim) \text{ and}$$

25  $Tmcom = T_{mmax}(Lim).$  --- (9B)

[Mode 2]

When  $Tdcomf < T_{mmax}(Lim) \leq$  small set value, the regenerative braking and the front and rear wheel hydraulic braking are employed as expressed by the  
30 following expressions (9C):

$$Tbcomf = Tdcomf - T_{mmax}(Lim),$$

$$Tbcomr = Tdcomr, \text{ and}$$

Tmcom=Tmmax(Lim). --- (9C)

[Mode 1]

When other case except the above Modes 4, 3 and 2,  
only the front and rear hydraulic braking is employed as  
5 expressed by the following expressions (9D):

Tbcomf=Tdcomf,

Tbcomr=Tdcomr, and

Tmcom=0. --- (9D)

[0063] At step S16 controller 24 calculates  
10 wheel-cylinder hydraulic pressure command values Pbcomf  
and Pbcomr for front and rear wheels using a constant K3  
on the basis of front and rear wheel hydraulic braking  
toque command values Tbcomf and Tbcomr, as follows.

Pbcomf = -(Tbcomf×K3)

15 Pbcomr = -(Tbcomr×K3) --- (10)

where K3 is the constant determined from the vehicle  
specifications previously stored in ROM.

[0064] At step S17 controller 24 calculates reference  
model response values Pbreff and Pbrefr of front and rear  
20 wheel-cylinder hydraulic command values Pbcomf and Pbcomr.  
The reference model response is a reference model  
response employed when hydraulic brake controller 18  
shown in Fig. 1 executes the feedback control of the  
front and rear wheel-cylinder hydraulic pressures. This  
25 reference model response is designed such that the  
hydraulic pressure actual value corresponds to the  
reference model response having a predetermined delay  
relative to the hydraulic command value. The reference  
model response may be designed by other method except for  
30 the feedback control method, and may be designed such  
that as a result a special responsibility relative to a

command value of a hydraulic servo system achieves the characteristic of the control system.

[0065] A reference characteristic  $F_{refb}(s)$  of the hydraulic servo system is, for example, represented by 5 the following expression (11).

$$F_{refb}(s) = 1/(T_{bref} \cdot s + 1) \quad \text{---(11)}$$

where  $T_{bref}$  is a time constant, characteristic  $F_{refb}(s)$  is obtained by discretizing the expression (11) as is the same manner discussed above, and reference model response 10 values  $P_{breff}$  and  $P_{brefr}$  for front and rear wheel-cylinder hydraulic command values  $P_{bcomf}$  and  $P_{bcomr}$  are also obtained by the same manner. Step S17 therefore corresponds to wheel-cylinder hydraulic reference model response value calculating means 35 in Fig. 2.

[0066] A hydraulic controller for wheel cylinders 2 in the first embodiment according to the present invention is constructed by a two-degree-of-freedom control system, and comprises a hydraulic control reference model 61, a feedforward compensator (phase compensator) 63, and a 20 feedback compensator 64, as shown in Fig. 9. Hydraulic control reference model 61 has a characteristic  $F_{refb}(s)$ . Feedforward compensator (phase compensator) 63 performs so as to correspond a response characteristic  $P_B(s)$  of a controlled object vehicle (wheel-cylinder hydraulic 25 control system) 62 to characteristic  $F_{refb}(s)$  of reference model 61.

[0067] Feedforward compensator 63 calculates a feedforward compensator quantity  $P_{ff}$  of the wheel-cylinder hydraulic pressure by filtering 30 wheel-cylinder hydraulic command value  $P_{bcom}$  through a characteristic  $G_{FF}(s)$  represented by the following expression (12). Herein,  $P_{bcom}$  is represented as a

common of front and rear wheel-cylinder hydraulic command values Pbcomf and Pbcomr.

$$G_{FF}(s) = Frefb(s)/P_B(s) \quad \text{---(12)}$$

[0068] In calculation of feedback compensation quantity Pfb using feedback compensator 64, reference model response value Pbref of the wheel-cylinder hydraulic pressure is first obtained by filtering wheel-cylinder hydraulic command value Pbcom through reference model 61 having a characteristic Frefb(s).

Then, wheel-cylinder hydraulic feedback difference  $\Delta P_b$  is obtained by subtracting an actual wheel-cylinder hydraulic pressure Pwc of controlled object (vehicle) 62 from wheel-cylinder hydraulic reference model response value Pbref as represented by the following expression (13).

$$\Delta P_b = Pbref - Pwc \quad \text{---(13)}$$

[0069] Subsequently, wheel-cylinder hydraulic feedback compensation quantity Pfb is obtained by filtering feedback difference  $\Delta P_b$  through a feedback compensator 64 having a characteristic  $G_{FB}(s)$  represented by the following expression (14).

$$G_{FB}(s) = (K_p \cdot s + K_i)/s \quad \text{---(14)}$$

In this embodiment, this characteristic is achieved by a basic PI controller, and therefore control constants  $K_p$  and  $K_i$  thereof are determined taking account of a gain margin and a phase margin.

[0070] Subsequently, a wheel-cylinder hydraulic control current Icom is obtained from the sum of feedforward compensation quantity Pff of wheel-cylinder hydraulic pressure and feedback compensation quantity Pfb,

and is outputted as a wheel-cylinder hydraulic control command to controlled object (vehicle) 62.

[0071] At step S18 controller 24 calculates front and rear wheel-cylinder hydraulic control errors  $\Delta P_{bf}$  and  $\Delta P_{br}$  from the following expressions. More specifically, controller 24 calculates front wheel-cylinder hydraulic control errors  $\Delta P_{bf}$  between front wheel-cylinder hydraulic reference model response value  $P_{breff}$  and front wheel-cylinder hydraulic pressure  $P_{wcf}$ , and rear wheel-cylinder hydraulic control errors  $\Delta P_{br}$  between rear wheel-cylinder hydraulic reference model response value  $P_{brefr}$  and rear wheel-cylinder hydraulic pressure  $P_{wcr}$ . In Fig. 1, actual front and rear wheel-cylinder hydraulic pressures  $P_{wcf}$  and  $P_{wcr}$  are represented as  $P_{wc}$ , and In Fig. 2, control errors  $\Delta P_{bf}$  and  $\Delta P_{br}$  are represented as  $\Delta P_b$ . Step S18 corresponds to hydraulic pressure control error calculating means 36 in Fig. 2.

$$\Delta P_{bf} = P_{breff} - P_{wcf}$$

$$\Delta P_{br} = P_{brefr} - P_{wcr} \quad \text{---(15)}$$

[0072] At step S19 controller 24 converts front and rear wheel-cylinder hydraulic control errors  $\Delta P_{bf}$  and  $\Delta P_{br}$  to the corresponding braking torque  $\Delta T_{bf}$  and  $\Delta T_{br}$ , respectively, using the following expressions (16).

$$\Delta T_{bf} = -(\Delta P_{bf} \div K_3)$$

25        $\Delta T_{br} = -(\Delta P_{br} \div K_3) \quad \text{---(16)}$

where  $K_3$  is a conversion coefficient based on the vehicle specification previously stored in ROM. Step S19 corresponds to torque converting means 37 in Fig. 2.

[0073] At step S20 controller 24 obtains the sum  $\Delta T_b$  30 of braking torque conversion values  $\Delta T_{brf}$  and  $\Delta T_{br}$  of the

hydraulic control errors from the following expression (17).

$$\Delta T_b = \Delta T_{bf} + \Delta T_{br} \quad \text{---(17)}$$

[0074] Further, controller 24 obtains regenerative braking torque correction quantity  $\Delta T_m$  by compensating the summed braking torque conversion value  $\Delta T_b$  using a phase advance compensation  $G_{ph}(s)$  represented by the following expression (18).

$$G_{ph}(s) = (T_m \cdot s + 1) / (T_{ph} \cdot s + 1) \quad \text{---(18)}$$

where  $T_m$  is a time constant, and  $T_{ph}$  is a time constant where  $T_{ph} \ll T_m$ . Step S20 corresponds to phase advance compensating means 38 in Fig. 2.

[0075] Time constant  $T_m$  of the response characteristic in the regenerative braking torque control system is processed in motor torque controller 21 so as to be brought to time constant  $T_{bref}$  in reference model 61 of the hydraulic braking torque control system. That is, the reference model employed in this control has a characteristic  $F_{refm}$  as same as the characteristic of the hydraulic braking torque control system. This arrangement is executed in view that when the hydraulic braking torque control system and the regenerative braking torque control system accurately follows the reference model without generating a response error, the total braking torque is always brought (adjusted) to the command value even if the total braking-torque command value or the braking torque distribution is varied.

[0076] At step S21 controller 24 obtains corrected regenerative braking-torque command value  $T_{mcom'}$  by correcting regenerative braking-torque command value

Tmcom by regenerative braking-torque correction quantity  $\Delta T_m$  as represented by the following expression (19).

$$T_{mcom}' = T_{mcom} + \Delta T_m \quad \text{---(19)}$$

Step S21 corresponds to regenerative braking torque  
5 correcting means 39 in Fig. 2.

[0077] At step S22 controller 24 outputs corrected  
regenerative braking-torque command value  $T_{mcom}'$  to motor  
torque controller 21 and front and rear wheel-cylinder  
hydraulic command values  $P_{bcomf}$  and  $P_{bcomr}$  to hydraulic  
10 brake controller 18.

[0078] Motor torque controller 21 controls motor 4  
through inverter 22 to bring the actual regenerative  
torque closer to corrected regenerative braking torque  
 $T_{mcom}'$ . Hydraulic brake controller 18 controls solenoid  
15 values 13, 14 and 16 to bring the actual front and rear  
hydraulic braking torques closer to front and rear  
wheel-cylinder hydraulic command values  $P_{bcomf}$  and  $P_{bcomr}$ ,  
respectively.

[0079] The hybrid brake system employed in the first  
20 embodiment comprises the regenerative braking apparatus  
which is relatively superior in control responsibility  
and the hydraulic braking apparatus which is relatively  
inferior in control responsibility. As shown in Fig. 2,  
the coordinated brake control system of the hybrid brake  
25 system comprises: a total braking torque determining  
means (total braking force calculating means) 31 which  
determines total braking force (torque) command value  
 $T_{dcom}$  according to the vehicle operating condition; a  
braking torque distributing means (braking force command  
30 value distributing means) 33 which distributes the total  
braking force command value  $T_{dcom}$  to regenerative  
braking-torque command value  $T_{mcom}$  and the hydraulic

braking-torque command value  $T_{bcm}$ ; a reference model response value calculating means 35 which calculates a reference model response value  $P_{bre}$  relative to a wheel-cylinder hydraulic pressure (braking force) command value  $P_{bcom}$ , on the basis of a braking force reference model taking account of a delay of the actual hydraulic pressure (actual braking force) at the hydraulic control system relative to the wheel-cylinder hydraulic pressure (braking force) command value  $P_{bcom}$ ; and a regenerative 5 braking force command value correcting means 39 which corrects the regenerative braking-torque command value  $T_{mcom}$  according to the braking force control error  $\Delta P_b$  ( $\Delta T_m$ ) between the actual braking force estimated value  $P_{wc}$  and the reference model response value  $P_{bre}$ ; wherein 10 the corrected regenerative braking-torque command value  $T_{mcom}'$  is employed in the control of the regenerative 15 braking apparatus.

[0080] With this arrangement according to the first embodiment of the present invention, in the correction of 20 regenerative braking-torque command value  $T_{mcom}$  for the regenerative brake apparatus performing a high control responsibility, regenerative braking-torque command torque  $T_{mcom}$  is corrected only by  $\Delta T_m$  which varies according to the braking force control error  $\Delta P_b$  between 25 the actual braking force estimated value (actual braking force estimated value  $P_{wc}$ ) relating to the hydraulic brake apparatus and the braking force command value (reference model response value  $P_{bre}$ ) relating to the hydraulic braking apparatus.

30 [0081] Accordingly, it becomes possible to prevent the excessive correction which has been executed in the case that the braking force command value of the

high-responsibility brake apparatus is corrected on the basis of a difference between a command value and an actual value of the low-responsibility brake apparatus as in a related coordinated braking control system. This  
5 solves a problem that the total braking force actual value largely deviates from the total braking force command value due to the execution of such an excessive correction.

[0082] More specifically, by setting the basic  
10 distribution to the hydraulic braking torque and the regenerative braking torque so as to bring the total braking torque actual value closer to the total braking-torque command value when the wheel-cylinder hydraulic pressure is accurately controlled according to  
15 the reference model response of the hydraulic control system, even if a difference is generated between the wheel-cylinder hydraulic pressure actual value and the reference model response value of the wheel-cylinder hydraulic pressure is generated by a large response delay,  
20 an overshoot or a stationary difference, the difference is covered with the regenerative braking torque.  
Therefore, it is possible to bring the total braking torque actual value closer to the total braking-torque command value. This advantage is also achieved in case  
25 that a braking operation quantity generated by a driver is varied, or in case that the distribution of the regenerative braking torque and the hydraulic braking torque is varied according to the vehicle operating condition.  
30 [0083] Referring Figs 10A through 13F, there are discussed the advantages of the coordinated brake control

system according to the first embodiment of the present invention.

[0084] Figs. 10A through 10F are time charts showing a simulation result in case that wheel-cylinder hydraulic control error  $\Delta P_b$  (braking force control error) is around zero during the control after master-cylinder hydraulic pressure  $P_{mc}$  is raised up at moment  $t_1$ .

[0085] Figs. 11A through 11F are time charts showing a simulation result in case that the correction of the regenerative braking-torque command value by regenerative braking torque correcting means 39 shown in Fig. 2 and the phase advance compensation by phase advance compensating means 38 in Fig. 2 are not executed even when wheel-cylinder hydraulic control error  $\Delta P_b$  (braking force control error) is generated at moment  $t_3$  during the control after master-cylinder hydraulic pressure  $P_{mc}$  is raised up at moment  $t_1$ .

[0086] Figs. 12A through 12F are time charts showing a simulation result in case that the correction of the regenerative braking-torque command value by regenerative braking torque correcting means 39 shown in Fig. 2 is executed, and the phase advance compensation by phase advance compensating means 38 in Fig. 2 is not executed when wheel-cylinder hydraulic control error  $\Delta P_b$  (braking force control error) is generated at moment  $t_3$  during the control after master-cylinder hydraulic pressure  $P_{mc}$  is raised up at moment  $t_1$ .

[0087] Figs. 13A through 13F are time charts showing a simulation result in case that both of the correction of the regenerative braking-torque command value by regenerative braking torque correcting means 39 shown in Fig. 2 and the phase advance compensation by phase

advance compensating means 38 in Fig. 2 are executed when wheel-cylinder hydraulic control error  $\Delta P_b$  (braking force control error) is generated at moment  $t_3$  during the control after master-cylinder hydraulic pressure  $P_{mc}$  is raised up at moment  $t_1$ .

[0088] The simulations shown in Figs. 10A through 13F were executed under the same condition. The condition is that the same constant-force braking from a predetermined vehicle speed is executed, that the regenerative coordinating brake control is started just after a moment  $t_1$  when a brake pedal is depressed, that during a low-speed traveling after moment  $t_2$  at when vehicle speed VSP is lowered to a set vehicle speed, the regenerative braking torque is gradually decreased as vehicle speed VSP is lowered and the wheel-cylinder hydraulic pressures of the front wheels or rear wheels are increased by the decreased quantity of the regenerative braking torque so as to correspond the total braking torque to the total braking-torque command value, and that the regenerative coordinated braking control is stopped just before the vehicle stops. Further, as discussed above, the total braking-torque command value is corrected so as to eliminate the influence of the disturbance by employing the deceleration feedback control in the first embodiment.

[0089] When the hydraulic control system idealistically functions, the vehicle deceleration (a deceleration of the driving wheels) is controlled at a constant value in response to the depression condition of the brake pedal as shown in Figs. 10A through 10F, particularly in Fig. 10B. The time charts in Figs. 10A through 10F are compared with the time charts in Figs. 11A through 13F in view of the operation at the time when

hydraulic control error  $\Delta P_b$  (braking force control error) is generated.

[0090] Figs. 11A through 13F basically show a case that the actual value of the front wheel-cylinder hydraulic pressure is momentarily and largely lowered relative to the command value due to the generation of a trouble in the front wheel brake hydraulic control system at moment t3 in a transient period after moment t2.

[0091] If neither the correction of the regenerative braking-torque command value through means 39 nor the phase advance compensation through means 38 is executed under the above-explained case as is the case represented by Figs. 11A through 11F, the control depends on only the feedback deceleration. Therefore, the compensation control cannot follow the actual change due to the response speed of the deceleration feedback control, and the deceleration of the driving wheel (front wheel) is momentarily and largely decreased in magnitude as shown in Fig. 11B. The reason thereof is that the error of the wheel cylinder 2 has a relatively large time lag in actually generating the driving wheel deceleration.

[0092] Figs. 12A through 12F show time charts in case that corrected regenerative braking-torque command value  $T_{mcom'}$  is varied by executing the correction of the regenerative braking-torque command value using the means 39 in Fig. 2, even if the front wheel brake hydraulic control system is put in a trouble state at moment t3.

[0093] At moment t3 when the actual value of the front wheel-cylinder hydraulic pressure is momentarily and largely lowered due to the trouble of the front wheel braking hydraulic control system, the hydraulic control error is rapidly compensated by the corrected

regenerative braking-torque command value by converting the control error between the actual value and the command value into a error-correction torque and by adding the error-correction torque to the regenerative 5 braking-torque command value. Therefore, it becomes possible to largely suppress the fluctuation of the driving wheel deceleration at moment t3, as compared with the case in Figs. 11A through 11F.

[0094] Figs. 13A through 13F show time charts in case 10 that corrected regenerative braking-torque command value  $T_{mcom}'$  is varied by executing the correction of the regenerative braking-torque command value using the means 39 in Fig. 2 and that the phase advance compensation is executed using the means 38 in Fig. 3, even if the front 15 wheel brake hydraulic control system is put in a trouble state at moment t3.

[0095] In this case of executing both of the 20 correction of the regenerative braking-torque command value and the phase advance compensation, the phase advance compensation functions to extremely decrease the delay of the regenerative braking torque control system relative to the command value. Therefore, the influence of the wheel-cylinder hydraulic pressure control error to the total braking torque is compensated by correcting the 25 regenerative braking torque as possible, and consequently it becomes possible to further largely suppress the fluctuation of the driving wheel deceleration at moment t3, as compared with the case shown in Figs. 12A through 12F.

[0096] Furthermore, the first embodiment according to 30 the present invention is arranged such that the regenerative braking-torque command value is determined

within maximum braking torque limit value  $T_{mmax}(Lim)$  obtained by limiting the allowable maximum regenerative braking torque  $T_{mmax}$  by the predetermined value with reference to the relationship shown in Fig. 7, at step 5 S14 before the total braking-torque command value is distributed into regenerative braking-torque command value  $T_{mcom}$  and hydraulic braking-torque command value  $T_{bcom}$  at step S15. Therefore, it becomes possible to ensure a margin for correcting regenerative 10 braking-torque command value  $T_{mcom}$  based on the wheel-cylinder hydraulic control error, and the advantage by this correction of regenerative braking-torque command value  $T_{mcom}$  is firmly ensured.

[0097] Further, the first embodiment according to the 15 present invention is arranged such that the regenerative brake apparatus is employed as a high-response braking means and a friction type brake apparatus is employed as a low-response braking means, and that the coordinated brake control system is adapted to the hybrid brake 20 system which is a combination of the regenerative brake apparatus and the friction type brake apparatus. Therefore, even if the friction type brake apparatus having a low-control-responsibility generates a transient control error, the regenerative brake apparatus having a 25 high-control-responsibility covers the shortage of the braking torque generated by the control error, and consequently the total braking-torque command value is achieved. Therefore, it becomes possible to prevent the driver from having the strange feeling during the 30 brake-pedal operating period.

[0098] Although the first embodiment according to the present invention has been shown and described such that

the friction type brake apparatus is employed as the hydraulic brake apparatus, it may not be limited to this, and there may be employed an electromotive brake system of pressing friction elements to a rotating disc or drum  
5 using an electric motor.

[0099] Referring to Figs. 14A and 17F, there is discussed a second embodiment of the coordinated brake control system according to the present invention. The basis construction of the second embodiment is basically  
10 the same as that of the first embodiment shown in Fig. 1.

[0100] The second embodiment is specifically arranged as shown in Fig. 14B such that a high-frequency component extracting means 40 is newly added between the means 37 and 38 of Fig. 2. A flowchart of Fig. 14A newly includes  
15 steps S20a and S20b instead of step S20 of the flowchart in Fig. 3. The other steps of the flowchart executed in the second embodiment are basically the same as steps shown in Fig. 3, and therefore the explanation of the other steps is omitted herein. The second embodiment is  
20 specifically arranged as shown in Fig. 14B such that a high-frequency component extracting means 40 is newly added between the means 37 and 38 of Fig. 2. As to steps S20a and S20b, the explanation is made as follows.

[0101] At step S20a in Fig. 14A subsequent to the  
25 execution of step S19, controller 24 extracts a high-frequency component from a total control-error torque conversion value  $\Delta T_b$  ( $\Delta T_b = \Delta T_{bf} + \Delta T_{br}$ ) of the hydraulic control error by filtering total control-error torque conversion value  $\Delta T_b$  ( $\Delta T_b = \Delta T_{bf} + \Delta T_{br}$ ) through a  
30 bypass filter  $G_{hp}(s)$  represented by the following expression (20).

$$G_{hp}(s) = Th_{pf} \cdot s / (Th_{pf} \cdot s + 1) \quad \text{--- (20)}$$

where  $T_{hpf}$  is a time constant. Therefore, step S20a corresponds to high-frequency component extracting means 40 in Fig. 14B.

[0102] At step S20b subsequent to the execution of  
5 step S20a, controller 24 calculates regenerative braking torque correction value  $\Delta T_m$  by compensating the high-frequency component of total control-error torque conversion value  $\Delta T_b$  of the hydraulic control error using a phase advance compensator  $G_{ph}(s)$  represented by the  
10 expression (20). Therefore, step S20B corresponds to a high-frequency component extracting means 40 of Fig. 14B. After the execution of step S20b, the program proceeds to step S21.

[0103] With the thus arranged second embodiment  
15 according to the present invention, total braking-torque command value  $T_{dcom}$  is determined by the deceleration feedback control so as to bring the actual deceleration  $\alpha_v$  closer to target deceleration  $\alpha_{dem}$  according to the vehicle operating condition.

[0104] The high-frequency component of total control-error torque conversion value  $\Delta T_b$  of the hydraulic control error is obtained by filtering total control-error torque conversion value  $\Delta T_b$  through the bypass filter, and is employed in the correction of the  
25 regenerative braking torque without directly employing of total control-error torque conversion value  $\Delta T_b$ .

Therefore, the coordinated brake control system of the second embodiment according to the present invention also ensures the following advantages.

[0105] The second embodiment according to the present invention is arranged such that the regenerative braking

torque is corrected so as to be decreased when front and rear wheel-cylinder hydraulic control errors  $\Delta P_{bf}$  and  $\Delta P_{br}$  ( $\Delta P_b$  in Fig. 2) are generated such that front and rear wheel-cylinder hydraulic pressures  $P_{wcf}$  and  $P_{wcr}$  ( $P_{wc}$  in Fig. 1) detected by sensor 20 becomes greater than front and rear wheel-cylinder hydraulic reference model response values  $P_{breff}$  and  $P_{brefr}$  obtained at step S17.

[0106] Further, the second embodiment is arranged to employ only the high-frequency component of total control-error torque conversion value  $\Delta T_b$  in the correction of the regenerative braking torque, and therefore the correction is executed during a transient period where front and rear wheel-cylinder hydraulic control errors  $\Delta P_{bf}$  and  $\Delta P_{br}$  are varying, and is not executed during a steady state where front and rear wheel-cylinder hydraulic control errors  $\Delta P_{bf}$  and  $\Delta P_{br}$  are stable. This arrangement of the second embodiment according to the present invention therefore prevents the lowering correction of the regenerative braking torque from being continuously and excessively executed even when front and rear wheel-cylinder hydraulic control errors  $\Delta P_{bf}$  and  $\Delta P_{br}$  are stable. Consequently, it becomes possible to avoid a detrimental effect of lowering the energy efficiency.

[0107] Referring to Figs. 15A through 17F, there are discussed the advantages of the second embodiment according to the present invention. Figs. 15A through 17F show time charts in case that although master-cylinder hydraulic pressure  $P_{mc}$  is increased stepwise at a moment  $t_1$ , front and rear wheel-cylinder

actual hydraulic pressures  $P_{wcf}$  and  $P_{wcr}$  become steadily larger than front and rear wheel-cylinder hydraulic reference model response values  $P_{breff}$  and  $P_{brefr}$ , respectively, due to the trouble of the front and rear  
5 wheel brake system at a moment  $t_3$ .

[0108] More specifically, Figs. 15A through 15F show time charts in case that only the deceleration feedback control is executed. In other words, only the deceleration feedback control is executed when  
10 wheel-cylinder hydraulic control error  $\Delta P_b$  (braking force control error) is generated at moment  $t_3$  during the control after master-cylinder hydraulic pressure  $P_{mc}$  is raised up at moment  $t_1$ .

[0109] Figs. 16A through 16F show time charts in case  
15 that the regenerative braking torque is corrected by the hydraulic control error in addition to the execution of the deceleration feedback control. That is, the deceleration feedback control and the phase-advance compensation are executed when wheel-cylinder hydraulic  
20 control error  $\Delta P_b$  (braking force control error) is generated at moment  $t_3$  during the control after master-cylinder hydraulic pressure  $P_{mc}$  is raised up at moment  $t_1$ , in this case.

[0110] Further, Figs. 17A through 17F show time charts  
25 in case that the regenerative braking torque is corrected only by the high-frequency component of the hydraulic control error according to the embodiment of the present invention. That is, the deceleration feedback control, the phase-advance compensation and the high-pass  
30 filtering process of the hydraulic control error are executed when wheel-cylinder hydraulic control error  $\Delta P_b$  (braking force control error) is generated at moment  $t_3$

during the control after master-cylinder hydraulic pressure  $P_{mc}$  is raised up at moment  $t_1$ , in this case.

[0111] When only the deceleration feedback control is executed, the actual deceleration  $\alpha_v$  largely fluctuates at moment  $t_3$  as shown in Figs. 15A through 15F. Therefore, this fluctuation is converged to the command value  $\alpha_{dem}$  by executing the deceleration feedback control.

[0112] When the regenerative braking torque is corrected by the hydraulic control error in addition to the execution of the deceleration feedback control, the fluctuation of the actual deceleration  $\alpha_v$  is momentarily decreased at moment  $t_3$ , and this correction of the regenerative torque continuously decreases the absolute value of the regenerative braking torque after moment  $t_3$ , as shown in Figs. 16A through 16F. Therefore, as is apparent from the comparison with the case of Figs. 15A through 15F in regenerative quantity, the energy efficiency is lowered.

[0113] When the regenerative braking torque is corrected only by the high-frequency component of the hydraulic control error according to the embodiment of the present invention, the fluctuation of the actual deceleration  $\alpha_v$  at moment  $t_3$  is decreased by this correction as shown in Figs. 17A through 17F. Further, since this correction using the high-frequency component is executed only during the transient state just after moment  $t_3$ , the absolute value of the regenerative braking torque is quickly converged to the maximum regenerative braking torque  $T_{mmax(Lim)}$  so as to obtain the regenerative quantity which is generally the same as that

of Fig. 15F. Therefore it becomes possible to prevent the lowering of the energy efficiency.

[0114] Further, even if the deceleration is again fluctuated by not executing the correction of the 5 regenerative braking torque under the steady state, such a correction under the steady state is sufficiently corrected by the deceleration feedback control. This is easily understood from the phenomenon that during a period  $\Delta t$  from the starting moment  $t_3$  of the correction 10 of the deceleration fluctuation to a starting moment  $t_4$  of the limiting of the regenerative braking torque, the fluctuation of the deceleration does not generate a large difference.

[0115] Finally, since each of the embodiments of the 15 present invention is arranged to correct the regenerative braking torque by a feedforward method based on the hydraulic control error, there is caused no interference between the deceleration feedback control and the feedforward control, even if this feedforward method is 20 employed with the deceleration feedback control.

[0116] This application is based on prior Japanese Patent Applications Nos. 2003-57299 and 2002-268794. The entire contents of the Japanese Patent Applications No. 2003-57299 with a filing date of March 4, 2003 and No. 25 2002-268794 with a filing date of September 13, 2002 are hereby incorporated by reference.

[0117] Although the invention has been described above by reference to certain embodiments of the invention, the invention is not limited to the embodiments described 30 above. Modifications and variations of the embodiments described above will occur to those skilled in the art in

light of the above teachings. The scope of the invention is defined with reference to the following claims.